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REVERSIBLE HYDRAULIC MACHINES (PUMP TURBINES) IN A POWER SYSTEM

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/Figures referred to are appended./

With the increase of power in power systems and of the consumer share of the load, the gap between the maximum, average, and minimum value of required power continues to widen. One way of shortening this gap is to use pumped-storage hydroelectric stations in the power system.

The most compact, simple and, hence, the cheapest solution is to equip pumped-storage hydroelectric stations with reversible units, i.e., a hydraulic machine which can operate alternately as a turbine or as a pump and an electric machine operating, respectively, as a generator or as a motor.

We will consider only the problem of equipping low-head hydroelectric stations capable of being used for water-storage with reversible hydraulic machines of the propeller type.

Reversibility of Hydraulic Machines of the Reaction Type

In principle, there is no difference in the operation of the blade wheel of a turbine and a pump, since the same law governs the operation of both types of machines. For equal size, rpm, and discharge through the runner, a turbine should use the same head as that produced by a pump when the efficiency is equal to unity. In other words, pump and trubine wheels are dynamically reversible. But since the efficiency is always less than unity, complete reversibility of the machine does not hold.

When a reversible unit operates between the same upper and lower waters i.e., when the difference between them is constant, losses in the canals up to and beyond the hydraulic machines are of predominant importance. They

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increase the head which must be developed by the runner when operating as a pump and decrease the head which is actually used by the wheel when operating as a turbine.

Experimental Data

In the Canal imeni Moscow, high-power adjustable-blade propeller pumps of the PV-250 type with a runner 2.5 m in diameter operate under a rated head of 8.5 m and a discharge of 30 cu m/sec. In 1946, Engineer Rumyantsev published the results of research work by canal engineers on one of the pumps which had been operated as a turbine. Without making any changes in the hydraulic part of the machine, the engineers let the water flow through the pump from the upper to the lower header. The pump was operated as a turbine, and the motor as a generator. The results of these experiments are shown in Figure 1.

The maximum effeciency obtained by operating the machine as a turbine, was 80%, while its maximum efficiency as a pump was 85%.

Previously, a group headed by Professor I. N. Voznesenskiy had made tests in the Hydromachines Laboratory of the Leningrad Polytechnic Institute imeni I. N. Smirnov of the reversibility of propeller wheels of pumps and turbines. Studies were made of the runner of propeller pump No 69 MVS, 350 mm in diameter, with its gates, flume, and draft tube, and of a turbine wheel which had the same cylindrical cross section as the pump wheel. The turbine runner (PVK-2) was tested with the conventional gates for axial turbines and a direct flaring vertical draft tube. The diameter of the runner model was 250 mm.

The discharge and efficiency of both runners coincided when the efficiency was high, thus indicating the dynamic reversibility of the axial wheel. The deviation of the results for model speeds (the optimum specific speed was $n'_1 = 160$ rpm for the pump runner and 112.5 rpm for the turbine runner) showed that different speeds must be used for pump and turbine operation and, naturally, different directions of rotation.

We use one result obtained in 1947 by A. G. Plotkina while investigating the operation of an axial pump in the Hydromachines Laboratory of the Leningrad Polytechnical Institute to estimate the influence of the type of flume and draft tube of machine in an axial reversible unit. Figure 2 is a diagram of this pump, Figure 3 the results of the tests made on it for runner-blade angles of 10° and 15°. The plus and minus signs indicate flow in different directions. The results obtained showed that in operation of casings without inserts, the discharge curves diverged widely (up to 400%). In operation with inserts, the total divergence was only 10%. If the flume and draft tube were made completely symmetrical, the curves might be expected to coincide.

This experimental data shows that it is definitely possible to build a propeller-type, reversible hydraulic machine. The efficiency obtained for the reversible machine will depend on correct design of the runner, flume, draft tube, and on the speeds selected for the machine for turbine and pump operation.

Cavitation in a Reversible Hydraulic Machine

A reversible hydraulic machine will operate under more severe cavitation conditions than the usual turbine, for the following reasons:

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1. The head generated in operation as a pump, as shown above, must be greater than the head used in turbine operation. This means that in pump operation the maximum speed at the blade increases, the minimum pressure decreases and, as a result, cavitation effects become more pronounced.

2. In the operation of a hydraulic machine, the draft head is a fractor influencing pitting. The presence of hydraulic resistance in the draft tube lowers the draft head in turbine operation and increases it during pump operation. Consequently, even if both the turbine and pump runners have identical cavitation properties and their geometric draft head is the same, the pump will still operate under more difficult conditions.

Both these factors tend to increase pitting of reversible hydraulic machines as compared with the ordinary turbine and thus the runner must be dropped closer to tailwater in the reversible machine.

Example of Reversible Unit Installation

Let us consider the feasibility of installing a reversible unit in a typical hydroelectric station. We take the following turbine parameters for this station:

Power	46,000	kw
Rated head	14.0	m
Discharge	375	cu m/sec
Rpm	75	
Runner diameter	8,000	mm
Max efficiency	92	%

Taking the worst possible type of hydroelectric station with a conventional turbine, we assume that the cavitation factor equals:

$$\sigma = \frac{H_a - H_s}{H} = \frac{10.3 - 1.25}{14.5} = 0.623 \qquad (1)$$

Assuming that the cavitation factor for a reversible machine is increased by roughlyy 0.15, we must drop the runner 2 m. Now the draft head H_s will be equal to -.75 m and thus

$$\sigma = \frac{10.3 + 0.75}{14.5} = 0.760, \qquad (2)$$

which requires that the runner have good cavitation properties.

Figure 4 shows the possible reversible unit in the hydroelectric station selected. The flume and draft tube are assumed to be symmetrical and to correspond in form to the draft tube of the conventional turbine (the bend in the head portion is assumed to be somewhat wider to compensate for the area occupied by the shaft casing).

The use of a flume of this form does not permit installation of the ususal gate equipment for turbines. The gate equipment must therefore be made conical (diagonal) with fixed vanes. This narrows the discharge characteristic of the machine but it will still have good regulation characteristics.

In accordance with the above calculations, the runner of the reversible machine is placed 2 m deeper than would be the case with the ordinary turbine of the station. Structurally it must be distinguished from the conventional type by special blades with sickle-shaped profiles.

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The same reference marks are employed in locating the electric machine (mortor-generator) as for the conventional turbine. Thus, the station buildings can be left unchanged.

Correspondingly, the base of the reversible machine unit is placed 1 m deeper. The same dimensions are found in the plan for a reversible machine unit as in the corresponding plan for a conventional turbine.

Th basic data for a water-storage unit in the hydroelectric station under consideration is taken from characteristics obtained by A. N. Smirnov. The runner diameter of the reversible hydraulic machine is assumed to be equal to the diameter of the station turbine, i.e., D=8,000 mm.

The efficiency of the full-scale unit is calculated from the formula

$$\eta_{f.s.} = 1-(1-\eta_m)\sqrt[5]{\frac{D_{f.s.}}{D_n}} \qquad (3)$$

and decreased 1.5-2% for turbine operation and 2-2.5% for pump operation. This was done because the tests which we used as a basis were made on units in which the runner-blade form was irreversible.

The values in connection with a change in the efficiency of the machine were calculated from the formulas

$$\frac{n_{im.}}{n_{if.s.}} = \sqrt{\frac{\eta_m}{\eta_{f.s.}}}; \qquad \frac{Q'_{im}}{Q_{if.s.}} = \sqrt{\frac{\eta_m}{\eta_{f.s.}}} \qquad (4)$$

The efficiency curves were plotted from the data given above for model tests converted by the formula

$$\eta_{f.s.} = \eta_m + [(\eta_{f.s.})_{max} - (\eta_m)_{max}] \qquad (5)$$

Consequently, the parameters of a reversible hydromachine for the hydroelectric station under consideration might have the values shown in the following tables:

Characteristics	For Turbine Operation	For Pump Operation	For Pump Operation	For Turbine Operation
Power (kw)	47,200	45,600	53,000	46,100
Head (m)	14	15	15	14
Delivery (cu m/sec)	375	260	310	375
Rpm	53.6	75.0	75.0	75.0
Diameter of runner (m)	8.0	8.0	8.0	8.0
No of runner blades	4	4	4	4
Assumed efficiency (%)	90.0	85.0	87.5	87.5

This data covers two types of operation of the unit, i.e., for different speeds in pump and turbine operation and for the same speed. In addition, two possible types of pump operation are considered: (1) for a required power equal to maximum turbine power; and (2) for discharge corresponding to maximum efficiency.

For operation as a turbine and in the optimum region of the characteristic curve, the parameters will be approximately:

Power (kw)	41,500	46,100
Rpm	53.6	75.0
Delivery (cu m/sec)	320	375
Head (m)	14	14
Assumed efficiency (%)	91	88.5

When the unit is operated as a pump (see table), the head is assumed to be roughly 1 m more than the head used by the turbine. Taking into consideration the comparative drop of tailrace level in pump operation, this increase in the head may be inadequate. Hence the possibility of a greater head for pump operation must be considered.

For the same 75 rpm as in the variation described above, the pump can operate under a head of 15.8 m with only a 1% lower maximum efficiency for the same discharge before.

The rpm was increased to evaluate pump operation with a higher head. At 83.3 rpm and a head of 17 m, the maximum efficiency was 88%, the discharge was 350 cu m/sec, and power was 63,000 kw. For a power of 46,000 kw, the maximum efficiency was 85% and the discharge was 235 cu m/sec.

All calculated types of pump operation are plotted on one graph (Figure 5). For comparison, efficiency and power curves for turbine operation with a K-91 type runner at 75 rpm and fixed gate opening are shown in the same graph.

Hence, the maximum efficiency for operation of a reversible machine as a turbine will be 91% at 53.6 rpm and 89% at 75 rpm. The maximum efficiency for operation as a pump will be 87.5% at 75 rpm.

Higher efficiencies can be obtained by designing special runners for reversible machines.

A reversible hydraulic machine will weight less than a conventional turbine of the same size. The work on gates and protective devices will be eased.

Instead of a swinging gate to cut off the upper water when the machine is shut down, high-speed roller gates spanning the upper canal must be erected.

On the negative side, it is apparent that installation of a reversible unit with symmetrical canals will entail a greater volume of concrete and earthwork.

Conclusions

The preliminary investigations cited above point up the following conclusions:

1. It is entirely possible to install a reversible hydromachine in newly constructed hydroelectric stations having the proper water-storage conditions.

2. Our turbine and electrical plants can build high-grade machines by using their experience and the results of experiments and theoretical works.

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Since the construction of pumped-storage hydroelectric stations is an imminent problem, our plants and planning and scientific institutions must now turn their attention to practical studies in designing the parts of reversible hydraulic machines.

3. The following problems should be among the first to receive through study:

a. Designing and testing the working parts of reversible hydraulic machines.

b. Designing high-power motor generators to rotate in either direction and, if possible, with different rpm.

c. Analyzing the practicability of using pumped-storage hydroelectric stations in various regions.

[Figures follow.]

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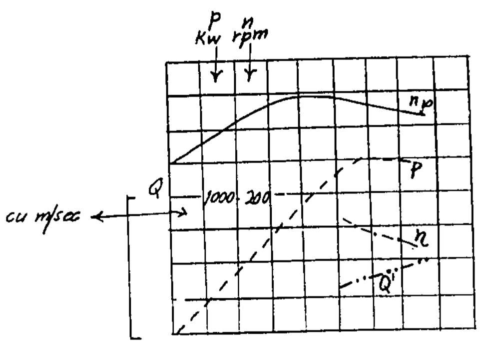

Figure 1. Curves for Operation of Machine in the Canal imeni Moscow as a Turbine
n_p is the Resonance rpm

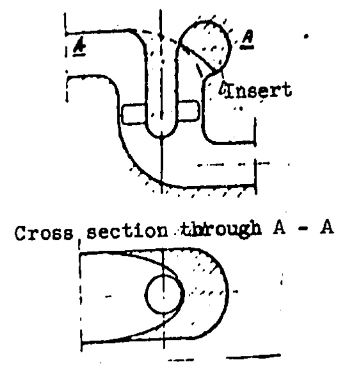

Figure 2. Plan of Canals for Reversible Axial Pump

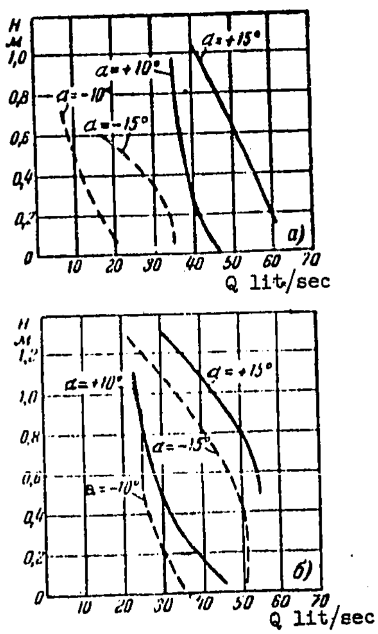

Figure 3. Results of Experiments with Reversbile Pump

a - without insert
b - with insert

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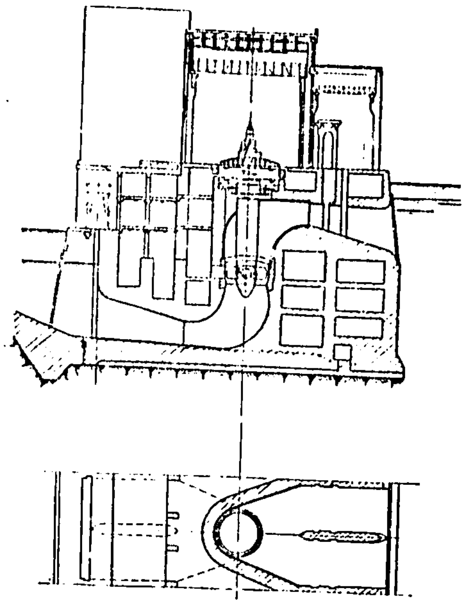

Figure 4. Plan of Reversible-Unit Installation

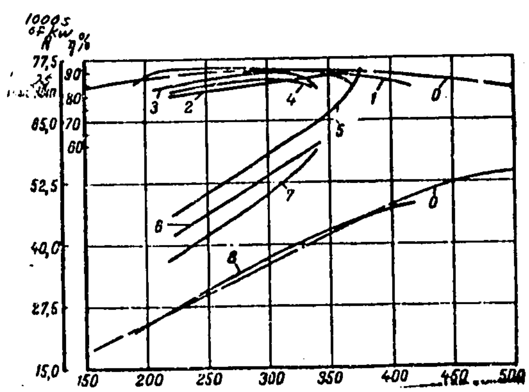

Figure 5. Operational Curves of Reversible Hydraulic Machine

1 - efficieny of turbine at 75.0 rpm, K=91, a_0 = const; 2 - efficiency of pump at 83.3 rpm, H=17 m; 3 - pump efficiency at 75.0 rpm, H=15 m; 4 - pump efficiency at 75.0 rpm, H=15.8 m; 5 - power of pump at 83.3 rpm, H=17 m; 6 - power of pump at 75.0 rpm, H=17 m; 7 - power of pump at 75.0 rpm, H=15.0 m; 8 - power of pump at 75.0 rpm, K=91, a_0=const; 0 - efficiency and power of turbine according to experiments at Leningrad Polytechnical Institute.

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